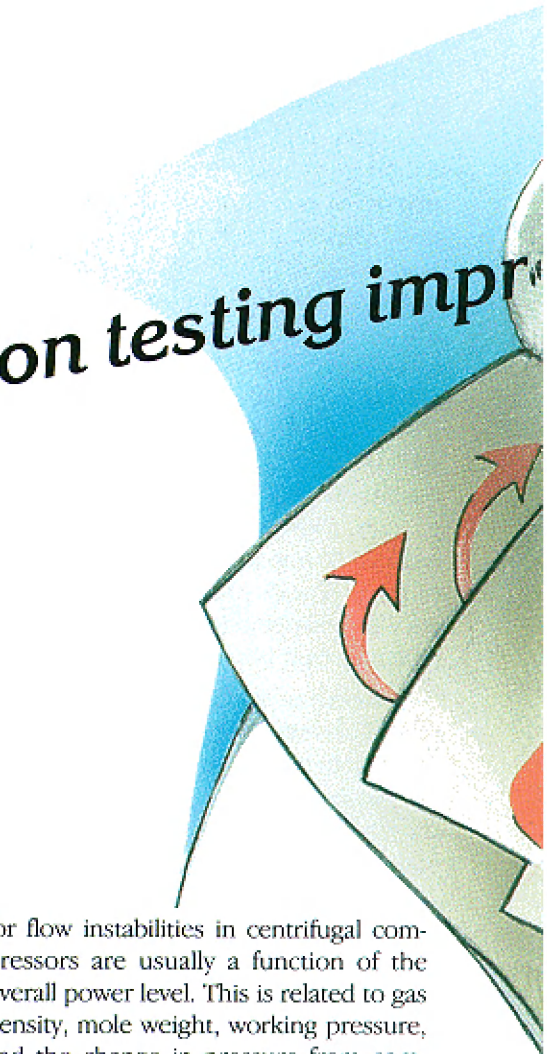
Machinery Messages

Case History

The Stall Zone

Compressor qualification testing impr



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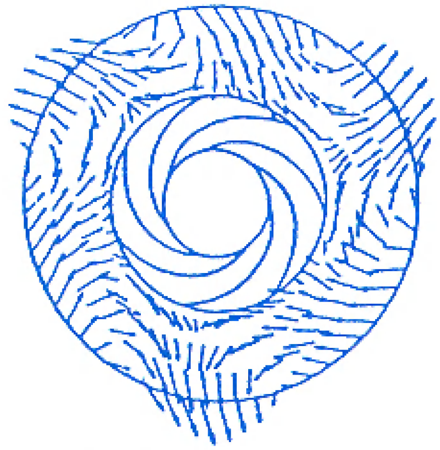
Bently Nevada's Machinery Diagnostics Services (MDS) group was recently contracted to acquire vibration data and perform diagnostics during qualification testing for a series of barrel-type, multi-stage, low pressure centrifugal compressors. The compressors are part of a gas injection process (repressurization) project. We tested each compressor according to applicable American Petroleum Institute and customer specifications.

These compressors employ eight stages in a straight-through design. They have modified inlet guide vanes and vaned diffusers and are coupled in a gas turbine - low pressure - high pressure compressor module. Squeeze-film damped bearings support the compressors at each end.

One low pressure compressor design experienced subsynchronous activity at less than rated (design) load during controlled test conditions. We had earlier ruled out bearing fluid-induced instabilities (fluid whirl and whip) as a potential source of the subsynchronous excitation. We closely regulated bearing and seal oil temperatures and pressures throughout all phases of the tests, and noted no discrepancies. The predominant subsynchronous vibration was evident at very low frequencies, lower than the 0.35X to 0.48X frequencies which are typical of bearing fluid whirl or whip. Therefore, we had to consider aerodynamic factors.

The compressor's discharge and suction end bearings were each fitted with a pair of radial proximity probes (with a redundant pair) to provide vibration information. The compressor's suction end had a thrust probe pair. As an additional diagnostic aid, the compressor was fitted with special pressure transducers at Stages 1 through 7, at the gas inlet loop and at the discharge spool. These dynamic differential pressure transducers were installed at the top of each return bend in the diffuser sections. They enabled us to observe the instantaneous, real-time pressure changes at each stage of the rotor and at the inlet loop (gas entrance passage).

The original mechanical qualification runs, conducted in the spring and summer of 1992, were without incident. During subsequent field qualification runs, the low pressure compressor had instability problems. The driving mechanisms for flow instabilities in centrifugal compressors are usually a function of the overall power level. This is related to gas density, mole weight, working pressure, and the change in pressure from compressor inlet to outlet (Δp). In the compressor industry, these flow instabilities are commonly called "surge" and "stall."

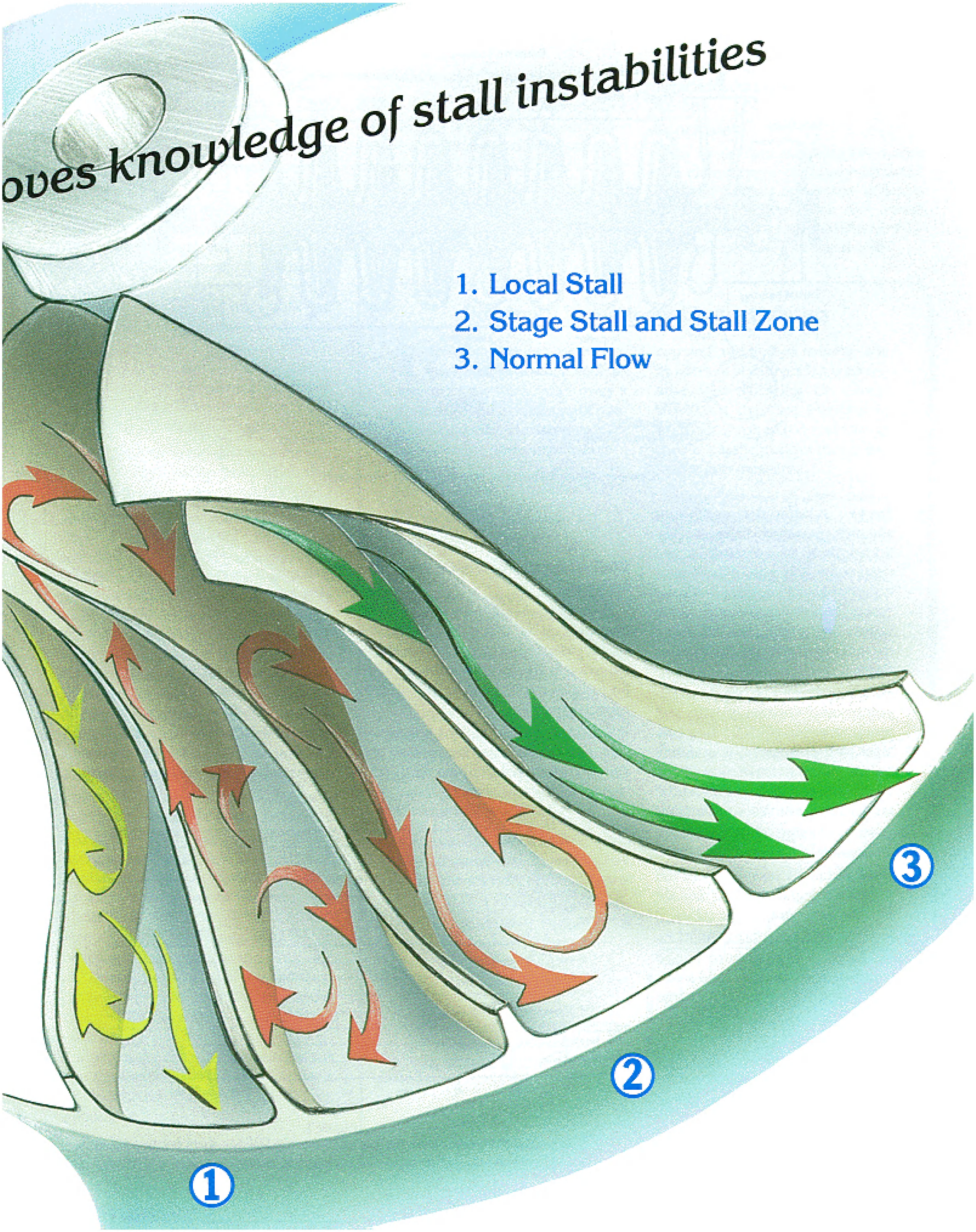


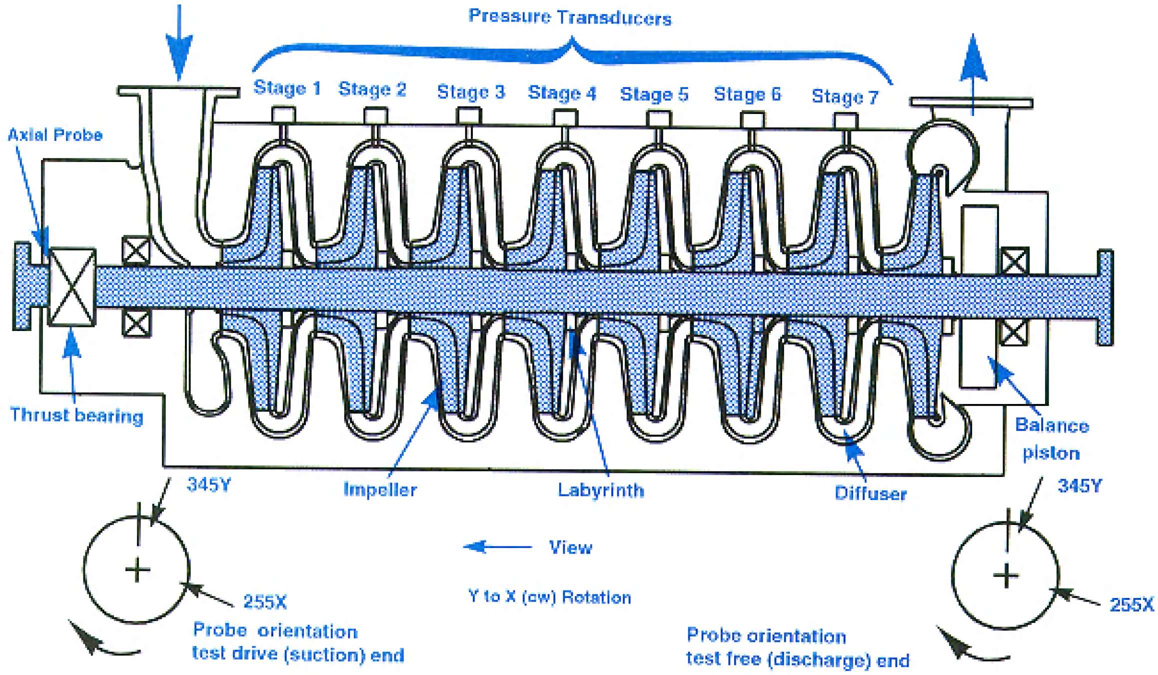
Example of Rotating Stall

A lot of documentation exists describing these performance problems. Generally, the following fluid dynamic phenomena can occur as a compressor's inlet gas flow is reduced. The following are common industry definitions:

- ***Local Stall*** - a flow separation or reversal in either an impeller or diffuser.
- ***Stage Stall*** - when a local stall increases to the point where one in a series of centrifugal impellers (and associated inlet and discharge stationary components) experiences reverse flow in part of its cross sectional flow area. The overall flow is still in the forward, pressurizing direction.
- ***Stall Zone*** - any cross-sectional area of an impeller or diffuser which undergoes a flow perturbation and which manifests symptoms of stall in a centrifugal compressor.

oves knowledge of stall instabilities





Compressor cross section, showing probe orientation

- ***Surge*** - periodic flow oscillations and pressure swings. If these oscillations include flow reversals, it is a deep surge. The rate of gas flow is not enough to adequately fill the spaces between consecutive guide vanes. This might also be called extreme stall. Surge propagates axially.
- ***Rotating Stall*** - also called "propagating stall." Consists of large stall zones, covering several blade passages, and propagates circumferentially at some fraction of rotor speed. The number of stall zones and the propagating rates vary considerably.

Both stall and surge are dangerous for the compressor, because:

- Rotor vibrations resulting from the streamline degradation can damage the labyrinths between the stages.
- Flow reversals can cause continuous increases in temperature at the impeller vane entrances (and a corresponding decrease in discharge head capability), resulting in surge cycling.
- Pressure variations between the intake and discharge ends can cause rapid changes in axial thrust, risking damage to the thrust bearing.
- Abrupt load and speed changes may occur. These can detrimentally affect the impellers and other internal compressor and driver components.

Stall onset and vibration response tracking

To better define the aerodynamic stall experienced in this compressor, we closely evaluated the pressure transducer information. We compared the pressure fluctuations at running speed to stall zone pressure changes.

We wanted to learn which stage or stages experienced stall first, relative to a specific flow coefficient, Q/N (the volumetric gas flow rate divided by operating speed). Q/N enables one to compare compressor flow rates over different operating speed ranges. We also wanted to determine the time it takes for a full stage stall to develop, the magnitude of the pressure change at that stage, and the corresponding vibration amplitudes.

We recorded vibration response and pressure transducer information on digital tape and later processed it with a Bently Nevada 108 Data Acquisition Instrument (108 DAI). We generated diagnostic plots with Bently Nevada's

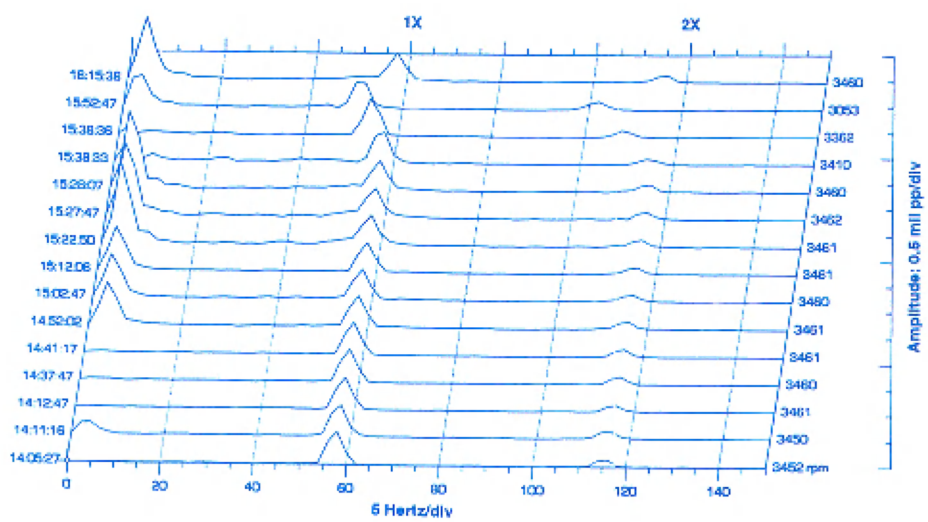


Figure 1
Spectrum waterfall plot from the free end horizontal probe, documenting the onset of stall and subsynchronous excitation at 4 Hz. Running Speed = 57.7 Hz.

ADRE® for Windows™ Software. ADRE® for Windows (Automated Diagnostics for Rotating Equipment) is powerful vibration diagnostic software that runs under Microsoft® Windows. Figures 1 through 4 show the time-mapped spectrum waterfalls of each horizontal and axial probe and the corresponding pressure fluctuations at Stage 4, from no stall to stage stall. The plots show that at 2:05 p.m., Stage 4 was the first stage to undergo increased pressure differences. This correlates well with the subsynchronous excitation we observed in the radial vibration response plots near that time.

Further pressure and vibration response tracking showed that, immediately after Stage 4 stalled, Stages 6 and 3 stalled simultaneously. Stages 5, 7, 2 and 1 followed, respectively. With reduced Q/N values, most of the inner stages of the compressor appeared to stall first. Also interesting is that Stages 3, 4, and 6 had identical mechanical designs (inlet guide vane angles, etc.), and were the first stages to suffer stall symptoms.

The greatest pressure fluctuation, from no stall to a full stall condition, was at Stage 6. Pressure increased by a factor of nearly four, from 0.26 bar to 1.0 bar (3.8 to 14.6 psi), and most of the other internal stages experienced stall zone pressure increases of at least 3:1. This pressure change is significant because it greatly affects mechanical stability. It excited a subsynchronous vibration component near 7% (4 Hz) of the running speed frequency (57.7 Hz), particularly high in amplitude at the free (discharge) end. When several stages stalled severely, the 4 Hz amplitude was as much as 38 μm (1.5 mils) pp uncompensated at the free end horizontal probe, almost three times higher than the running speed (1X) component! Because of the small internal clearances in this compressor, vibration in excess of 44 to 51 μm (1.75 to 2.00 mils) pp could damage the labyrinths and seals. We closely monitored vibration levels for any large increase, and were ready to shut down the compressor immediately. Drive end vibration was believed to be lower due to coupling effects. The free end could "float" on the test stand and take up as much of the allowable clearance as possible. It also appeared that the drive end stages were less susceptible to instantaneous stall.

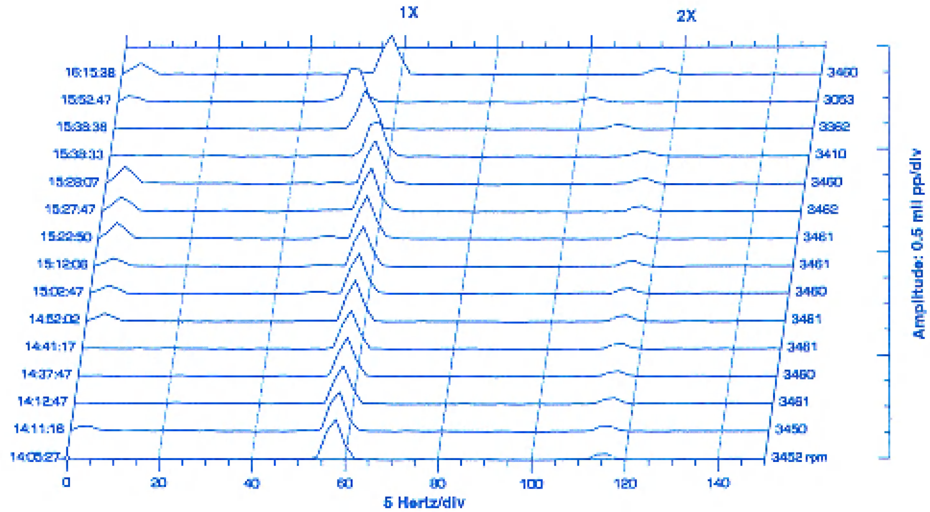


Figure 2
Spectrum waterfall plot from the drive end horizontal probe. Onset of stall and subsynchronous excitation at 4 Hz. Running Speed = 57.7 Hz.

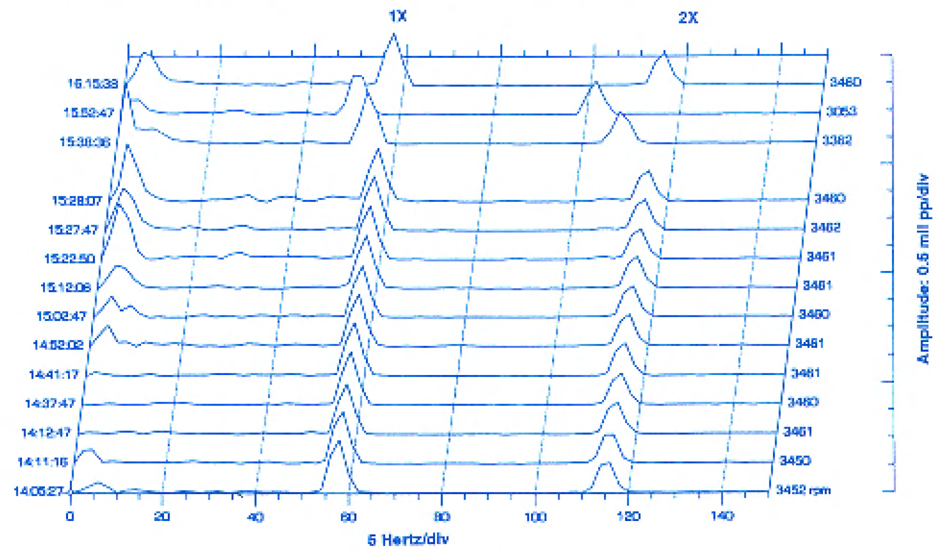


Figure 3
Thrust probe, documenting the onset of stall and subsynchronous excitation at 4 Hz. Running speed = 57.7 Hz.

This was probably due in part to the stabilizing effect of the balance piston. This behavior was not as predominant during later hydrocarbon (high horsepower and high pressure) testing, when the low pressure compressor was coupled to the turbine/compressor module.

The pressure gradient increase (the gas induced force) produced across one or more stages during stall appears to regulate system stability. Overall rotor stability, therefore, is closely governed by the effect of impeller and diffuser aerodynamic (gas kinetic) considerations and the stiffness relationships of the bearing, shaft and diaphragm system. Altering these two parameters, either by controlling inlet gas flow conditions or by changing the physical design of internal compressor components, will have a measurable impact on performance. Indeed, subsynchronous vibration intensity was very sensitive to design changes. Later compressor component redesigns proved this to be true. Subsequent modifications of the inlet guide vanes, balance piston configuration, diffusers, and labyrinths caused the vibration response to change. In addition to the predominantly low frequency vibration component typical of stall, further analysis of several design configurations revealed the presence of other subsynchronous components, including re-excitation of the rotor first balance resonance near 2800 cpm.

Orbit and timebase plots, made as the unit was experiencing stall, reveal some interesting behavior (Figures 5 and 6).

The plots show clockwise shaft rotation and vibration precession from no stall to surge. Note the violent response as the unit becomes unstable. A dramatic pulsation effect is evident at stall and surge in response to the gas' pressure gradient and sudden rotor acceleration and deceleration. From normal to surge conditions, vibration amplitude at the free end horizontal probe increased from 20 μm (0.80 mils) pp to 40 μm (1.59 mils) pp overall. Vibration amplitude at the drive end horizontal probe increased only marginally, from 21 μm (0.83 mils) pp to 25 μm (0.98 mils) pp. Stall symptoms were occurring well below the design flow point, at amplitudes greater than the customer-specified limits of 29.7 μm (1.17 mils) pp overall; therefore, the unit was disqualified. In several engineering tests conducted on similar impeller designs, with a fully developed stall, a 3:1 stage pressure change ratio corresponded with a 2:1 increase in overall vibration at the free end radial probes. We found that we could test the compressor during normal operating conditions and make a reasonable prediction of "worst case" vibration amplitude during stall, by keeping the mechanical configuration the same and testing the compressor at different speeds. The same comparison, under the same conditions, could not be made for the thrust or drive end radial probes.

While reducing gas flow, we observed that a stage can stall in as little as two to three seconds, depending on gas valve control. We repeated this behavior in each compressor stage and over several speed ranges. Finally, we backed the compressor out of stall (moved the operating point away from the surge line) to a higher flow coefficient or stability zone. We observed a pressure-related hysteresis effect on the instability. Once the unit had several stages stalling, it took a longer time to redevelop a stable flow than it took for the flow to become unstable.

The combined effect of stall flow anomalies in an impeller or diffuser also produces a marked "shift" in the compressor's performance curve (Figure 7). This reflects the reduced output from the individual stage. We observed this while plotting the flow/isentropic head data during testing.

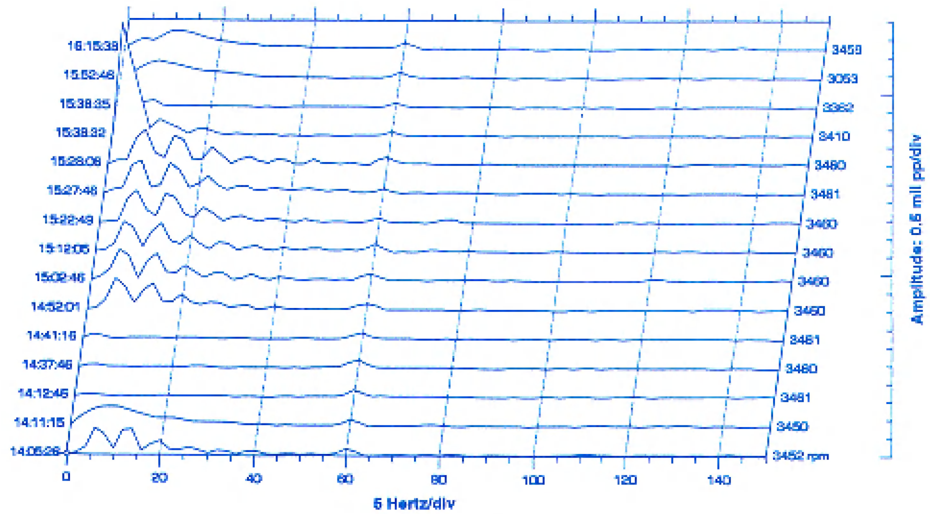


Figure 4
Stage 4 pressure transducer indicates the onset of stall. 1 mil pp = 0.3 bar (4 psi).

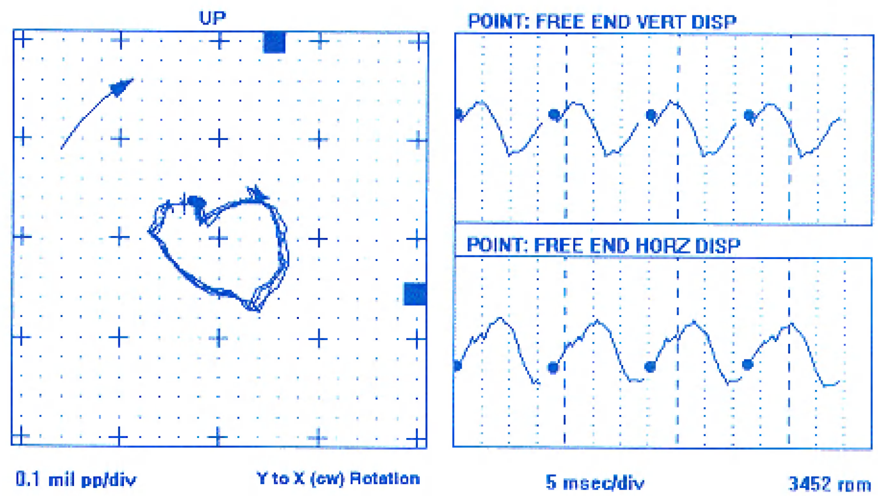


Figure 5
Free end orbit and timebase plots. Normal operation.

Conclusion

The OEM closely evaluated the relationships between the stall flow range, mechanical configuration, vibration characteristics, and design point Q/N location. They are using this information to create an optimum design and to increase their confidence in this unit's long term reliability. The distinction between stall, unique to an impeller or to a diffuser, was not easily defined, either in reference literature or in this project's results. However, we believe that some significant rotating stall was occurring prior to system surge. Severe stall can damage compressor and driver components.

During the performance portion of the qualification test, reducing the flow rate caused a reduction in the gas' mean velocity and fluid entry angle into a single stage. This significantly altered its Reynolds Number, kinetic energy, pressure ratio and surge margin, among other parameters. The stall cell formed at that particular location then became the chief driving mechanism of low frequency excitation, which we observed at near 7% of running speed for this design. Depending on the physical configuration of the compressor bearing, shaft and seal components, rotor system stiffness characteristics can also change. This can induce excitation of other subsynchronous vibra-

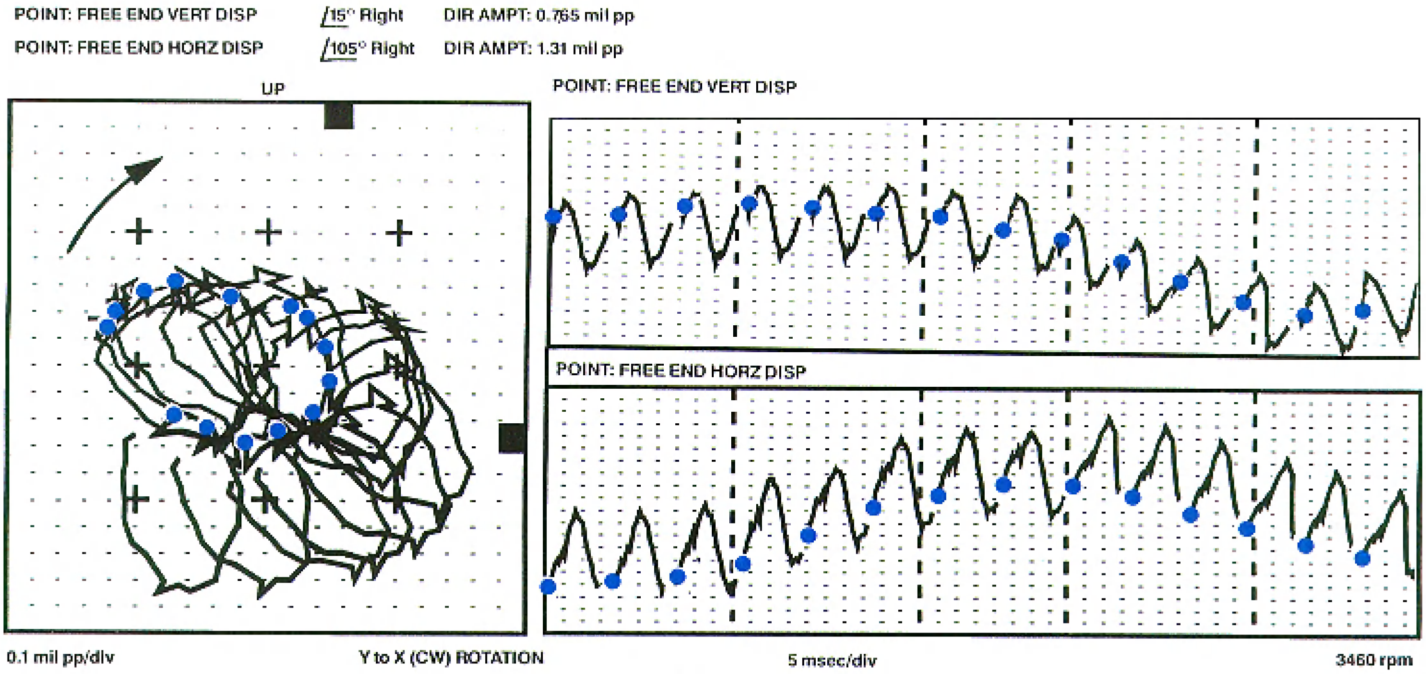


Figure 6
Orbit and timebase plots enhanced to illustrate the entire precession of the rotor in fourteen revolutions.
The Keyphasor® dots show the sampled form of the orbit and timebase plot of the low frequency component.

tion components, including those near the lateral balance resonance. This is particularly undesirable because it can greatly reduce the stability threshold.

During stall, pressure transducer information proved very valuable in understanding compressor rotor behavior relative to internal stage pressure and vibration changes. Evaluating the effects of changes to the compressor components would provide further insight into stall zone characteristics and their ability to act as low frequency driving mechanisms. Subsequent corrections by the OEM to the balance piston, labyrinths and eye (impeller) seals improved performance for various speed ranges and gas flow conditions.

For acquiring non-vibration data, process variable transducers can be connected to a Bently Nevada 208 Data Acquisition Interface Unit (DAIU) and properly scaled in the ADRE® For Windows Software. This allows you to directly acquire and read process variables, such as pressure, flow rate, etc., without converting scale factors. ADRE for Windows Software also allows you to easily retrieve, reduce and analyze archived ADRE 3 databases. This permits you to apply the many ADRE for Windows tools to previously created 108 DAI/ADRE 3 databases. ■

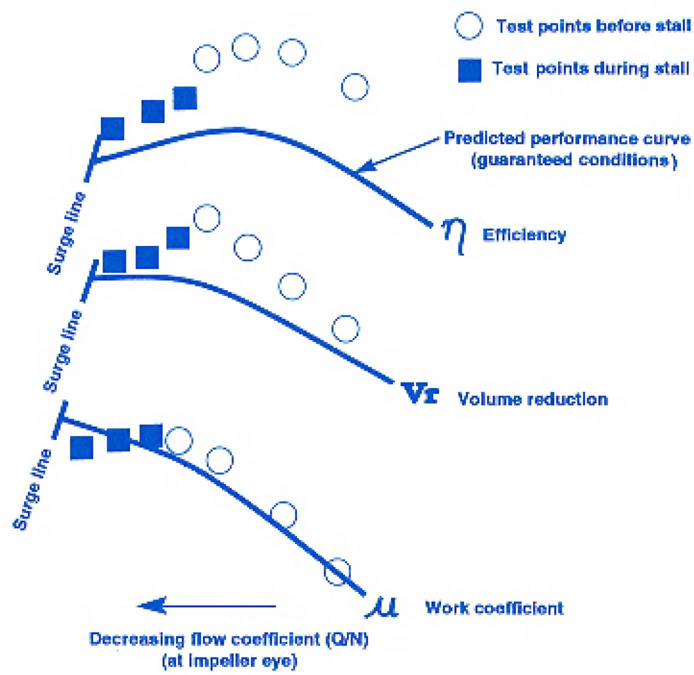


Figure 7
Compressor performance curve.

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